DETERMINATION OF

ELEVATED-TEMPERATURE FATIGUE DATA

ON REFRACTORY ALLOYS

IN ULTRA-HIGH VACUUM

FIRST QUARTERLY REPORT

TRW EQUIPMENT GROUP

THOMPSON RAMO WOOLDRIDGE INC.

CLEVELAND, OHIO

FIRST QUARTERLY REPORT

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DETERMINATION OF ELEVATED-TEMPERATURE FATIGUE DATA ON REFRACTORY ALLOYS IN ULTRA-HIGH VACUUM

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FOREWORD

The work described herein is being performed by Thompson Ramo Wooldridge Inc. under the sponsorship of the National Aeronautics and Space Administration under Contract NAS-3-6010. The purpose of this study is to obtain fatigue life data on refractory metal alloys for use in designing space power systems.

The program is administered for Thompson Ramo Wooldridge Inc. by E. A. Steigerwald, Program Manger. C. R. Honeycutt and J. C. Sawyer are the Principle Investigators.

ABSTRACT

A preliminary design for the vacuum fatigue test chambers has been developed in cooperation with the equipment vendor. This design, which is described briefly, makes extensive use of facilities already on hand for the vacuum creep program. In addition, a bench test set-up of the proposed design for the vibration loading system has been constructed and is in use to study problems of efficiency and reliability.

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INTRODUCTION

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The purpose of this investigation is to generate fatigue data for refractory alloys at elevated temperatures in ultra-high-vacuum environments. The ultimate objective of the program is to determine whether fatigue life or creep[1] is the limiting design parameter for refractory metal alloys in turbine applications.

At present, very little data exist concerning the fatigue properties of refractory metals and no design information is available at high frequencies in ultra-high vacuum. Since suitable equipment and methods for high-frequency fatigue testing in ultra-high vacuum at elevated temperatures are not available, the initial phase of the program deals with the development of test equipment which can vibrate specimens at 10 to 20 kilocycles per second at 1800 to 2200°F in a vacuum of 1×10^{-8} Torr. or better. In the second phase of the program, fatigue tests will be conducted on two selected refractory metal alloys.

This report describes the general design of the test equipment and presents current results on bench tests being made to study design problems in the high frequency vibration system. Author

VACUUM SYSTEM

A major part of the effort during this quarter has been devoted to designing and initiating procurement of four identical vacuum-fatigue test chambers. The system design is illustrated in Figure 1. Stainless steel is used for the system structure, all ports or flanges are sealed with OFHC copper gaskets, and bellows are used only where no vibration is present. In general, the vacuum chamber is composed of a spool piece[2], an ion pump, a dished front head, and a weight chamber for internal loading.

[1]Being studied on Contract NAS-3-2545.

[2]The cylindrical section which serves as the actual test chamber.

The spool piece is a double-walled, water-cooled cylinder, 22 inches in diameter with the axis horizontal. Feedthroughs for thermocouples, cooling water, electrical current for the furnace elements, electrical current for an internal light source, the mass analyzer flight tube, and the vacuum gauge are placed in the wall of the spool piece. A 500 liter/second integral ion pump is located at the back of the vacuum chamber along with a 6-inch access port.

The front closure is a double-walled, water-cooled, dished head with a nominal diameter of 22 inches to which is attached half of the furnace cold-wall and shield pack (but not the element). This arrangement permits rapid access to the furnace to facilitate changing specimens. The front closure contains a 4-inch diameter sightport for viewing the test specimen, a remotely operated shutter to protect the sightport from fogging when not in use, and cooling water feedthroughs for the front half of the furnace cold-wall.

The weight chamber below the furnace is also double-walled for water-cooling and will accommodate 350 pounds load. At the bottom of the extension chamber, a bellows is available for coupling the specimen to external dead-weight loads in the range 350-2000 pounds.

The split furnace design used in this system is shown in Figure 2. The front half of the trace-cooled copper cold-wall is attached to the front closure while the back half is rigidly mounted in the spool piece. Similarly, each half of the cold-wall contains half of the tantalum heat shield assembly. The tantalum heating element, which is also split, is supported by clamps attached to the high-current feedthroughs at the top of the spool piece. The back half of the element is held captive, while the front half is easily removable. This arrangement permits accurate positioning of the heating element around the test specimen before the system is closed.

The temperature control system for the fatigue apparatus will be identical to that used on the vacuum creep program[1]. In the fatigue studies extensive use is also being made of the following facilities from the creep program:

    1. roughing cart
    2. bakeout oven cart
    3. precision extensometer
    4. precision optical pyrometer

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[1]Contract NAS 3-2545

5. residual gas (mass) analyzer
6. recirculating cooling water installation, and
7. emergency electrical power installation.

An extensive description of this equipment has been previously presented (1, 2).

For the tension-tension fatigue stresses required in this program, dynamic loads are superimposed on a static load on the specimen. Static loads are produced by hanging the test specimen (in grips) from the adjustable top port assembly on the spool piece, and dead-weight loading the bottom grip assembly. Both the support at the top and the dead-weight loading support at the bottom must be located at nodal points of the superimposed dynamic load system. This is necessary to minimize power loss in the vibration system and to prevent vibration from reaching the bellows seals. Dynamic loads are applied to the test specimen through the top grips from a mechanically-resonant vibrating crystal and support structure mounted outside the vacuum chamber at the adjustable top port. This vibrating member, the specimen, and its grips must all be mechanically resonant at a particular vibration frequency in order for such a method of dynamic loading to be effective.

VIBRATION SYSTEM

Dynamic loading is applied to the specimen through the top port of the vacuum chamber. The resonant drive train shown in Figure 3 is used with nodes placed at both the top port seal and at the static-load attachment point. This vibration structure is driven at its resonant

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[1]J. C. Sawyer and C. H. Philleo; "Generation of Long Time Creep Data of Refractory Alloys at Elevated Temperatures," NASA Contract No. NAS 3-2545, 4th Quarterly Report, 1 July, 1964.

[2]J. C. Sawyer and E. B. Evans; "Generation of Long Time Creep Data of Refractory Alloys at Elevated Temperatures," NASA Contract No. NAS 3-2545, 3rd Quarterly Report, 20 April, 1964.

frequency by an externally mounted piezoelectric (ferroelectric) transducer. The following areas have been identified as critical design problems:

1. the shift in the resonant frequency of the structure as the specimen and its grips are heated;

2. the total input power required to fracture a specimen, considering losses in mechanical joints;

3. the lateral flexure in the drive train;

4. the optimum design for the top port seal to achieve maximum lateral compliance at the node located there.

In order to minimize or eliminate these problems, a bench test has been initiated with the apparatus shown in Figure 4. Results to date indicate that the calculation of the resonant frequency of cylindrical bars or stepped-horn type velocity transformers from static Young's modulus data result in some inaccuracy (See Table I). This difference is greatest for the titanium stepped horn. Such a result could come from a lack of precision in the value used for Young's modulus or from geometric effects in the stepped horn. Additional tests will be conducted to resolve this problem.

Thus far, tests of crystal and velocity transformer combinations have been made without matching the resonant frequencies of the two components. Peak-to-peak mechanical deflections of 0.11 mils $(1.1 \times 10^{-4}$ in.) have been measured at 15,795 cps with a 6-inch long stepped horn, and at 0.20 mils $(2 \times 10^{-4}$ in.) at 17,670 cps with a 5-1/4-inch long stepped horn. Improvements in the efficiency of the electrical coupling between the audio amplifier and the PZT crystal, and matching the resonant frequencies of the crystal and stepped horn can both be expected to increase the output mechanical vibration.

TABLE I

DIFFERENCE BETWEEN OBSERVED AND CALCULATED[a]

RESONANT FREQUENCIES

Material	Resonant Frequency (cps)		Difference[b] (%)
	Observed	Calculated	
Titanium - commercially pure (stepped-horn)	17, 460	19, 000	+ 8. 9
Columbium - commercially pure (cylindrical bar)	11, 720	11, 610	- 0. 9

[a]For the calculation of resonant frequencies, see the appendix.

[b]$\dfrac{\text{Calculated-Observed}}{\text{Observed}}$ X 100%

WORK IN PROGRESS

Vacuum System

The final vacuum system design will be reviewed and released for manufacture. The procurement of associated components such as a recorder, controller, ion pump power supplies, etc., will also begin. Shipment by the vendor of two of the completed systems is scheduled for 12 January and with the remaining two being delivered on 19 January, 1965.

Vibration System

Experimentation with the bench set-up will continue toward the aim of eliminating or minimizing the design problems outlined earlier in this report. Adoption of a final design is scheduled for 1 December, 1964.

Vacuum Fatigue Tests

Proof tests of the vacuum fatigue facility are scheduled for completion by 1 March, 1965. Fatigue tests of refractory metal turbine alloys will commence on that date.
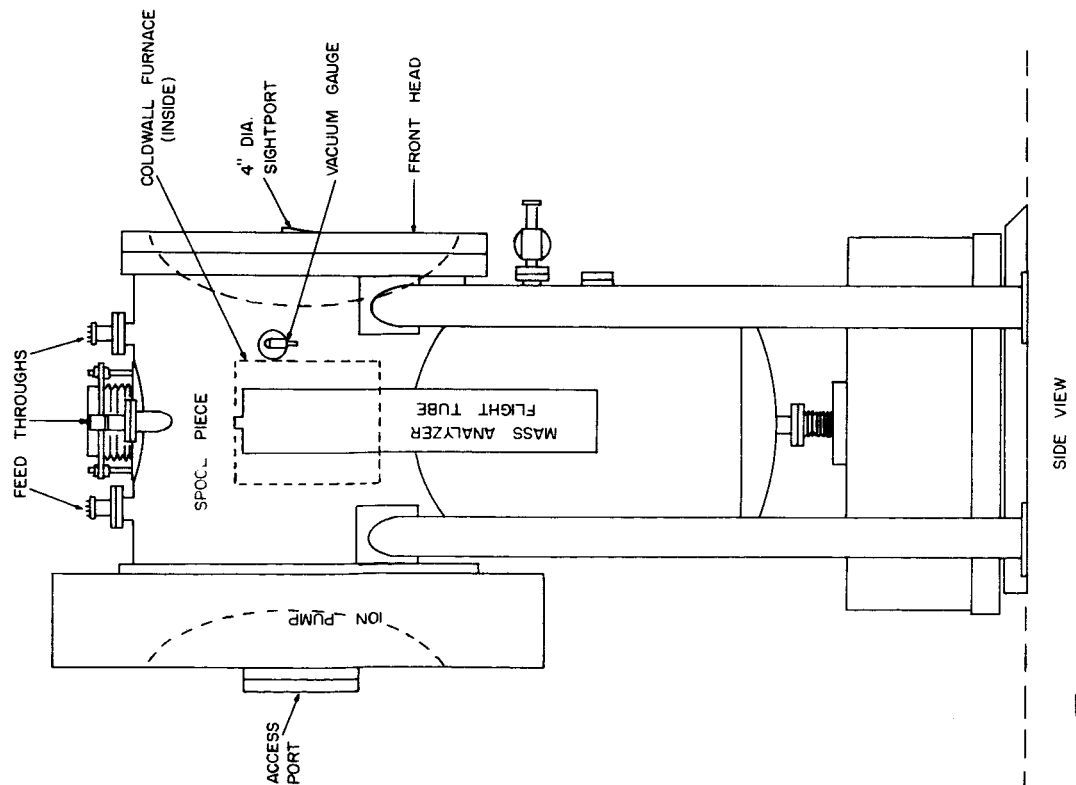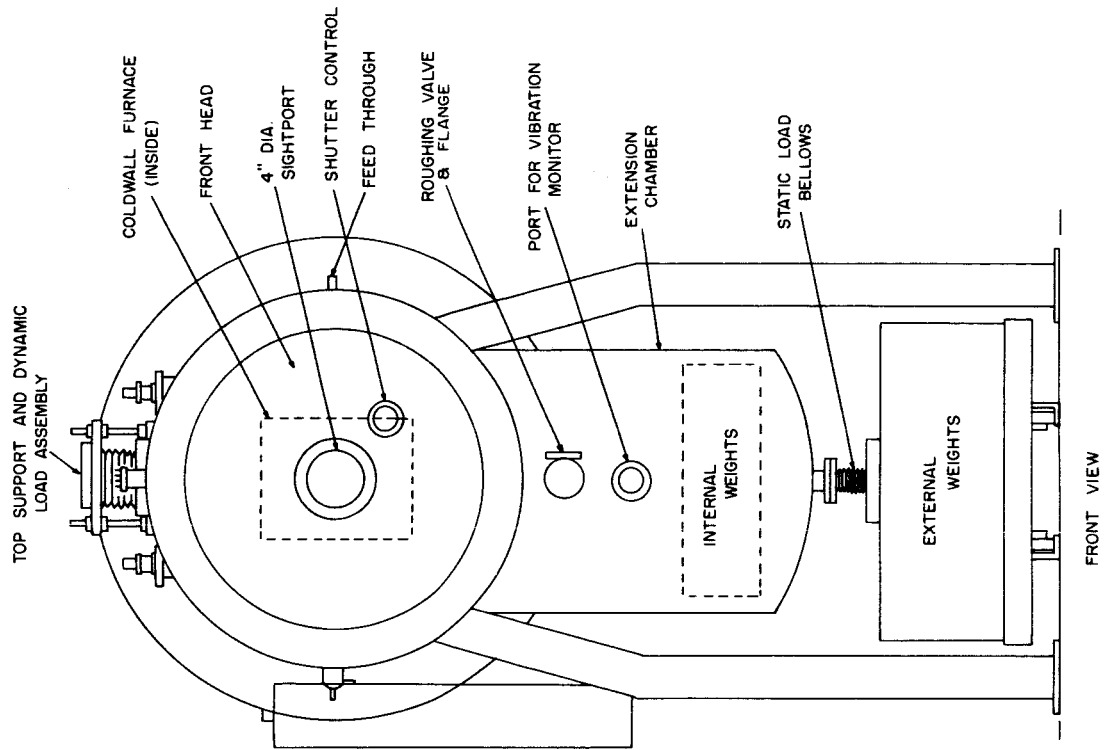
VACUUM FATIGUE SYSTEM

FIG. I

6546

PERMANENTLY MOUNTED
BACK HALF OF THE
COLDWALL

HEATING   ELEMENT
(BACK)   (FRONT)

REMOVABLE FRONT
HALF OF THE
COLDWALL PLUS
HALF THE HEAT
SHIELD ASSEMBLY

HALF OF THE
HEAT SHIELD
ASSEMBLY

SPLIT   COLDWALL   FURNACE

(NOTE: HEATING ELEMENT TABS AND FURNACE
ASSEMBLY GUIDE MECHANISM ARE NOT SHOWN)

PZT TRANSDUCER

TOP PORT FLANGE

Ti VELOCITY TRANSFORMER

VERTICAL SUPPORT

Cb VELOCITY TRANSFORMER
& TOP GRIP

SPECIMEN

Cb BOTTOM GRIP

NODE

STATIC LOAD   STATIC LOAD

VIBRATION MONITOR
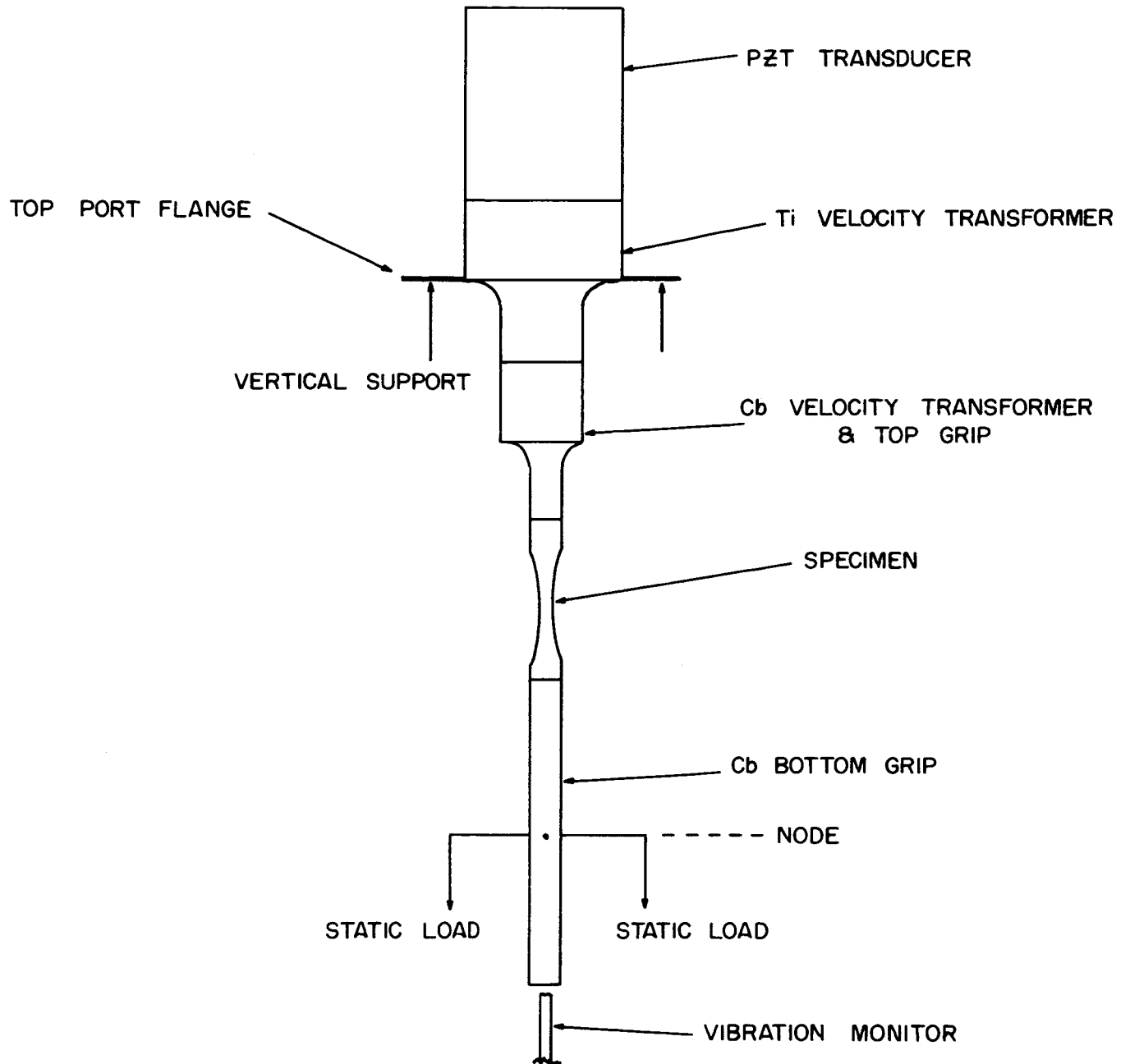
RESONANT VIBRATION DRIVE TRAIN

FIG. 3

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┌─────────────────────┐     ┌──────────────┐     ┌──────────────┐
│       Audio         │     │    Audio     │     │    Audio     │
│     Frequency       │ ──▶ │  Amplifier   │ ──▶ │   Coupling   │
│     Oscillator      │     │   60 watt    │     │ Transformer  │
│   600–27,000 cps    │     │              │     │   65 watt    │
└─────────────────────┘     └──────────────┘     └──────────────┘
          │                                              │
          ▼                        ┌──────────────┐      │
  ┌──────────────┐                 │     PZT      │ ◀────┘
  │ Counter and  │                 │   Crystal    │
  │ Oscilloscope │                 │  Transducer  │
  └──────────────┘                 ├──────────────┤
          ▲                        │   Titanium   │
          │                        │   Velocity   │
          │                        │ Transformer  │
          │                        └──────────────┘
  ┌──────────────┐      ┌──────────────┐
  │  Microphone  │ ◀─── │ Fiber Optics │
  │    Pickup    │      │    Pickup    │
  └──────────────┘      └──────────────┘
                                │
  ┌──────────────┐      ┌──────────────┐
  │  Calibrated  │ ◀─── │   Fotonic    │
  │ Oscilloscope │      │    Sensor    │
  └──────────────┘      │   Control    │
                        └──────────────┘
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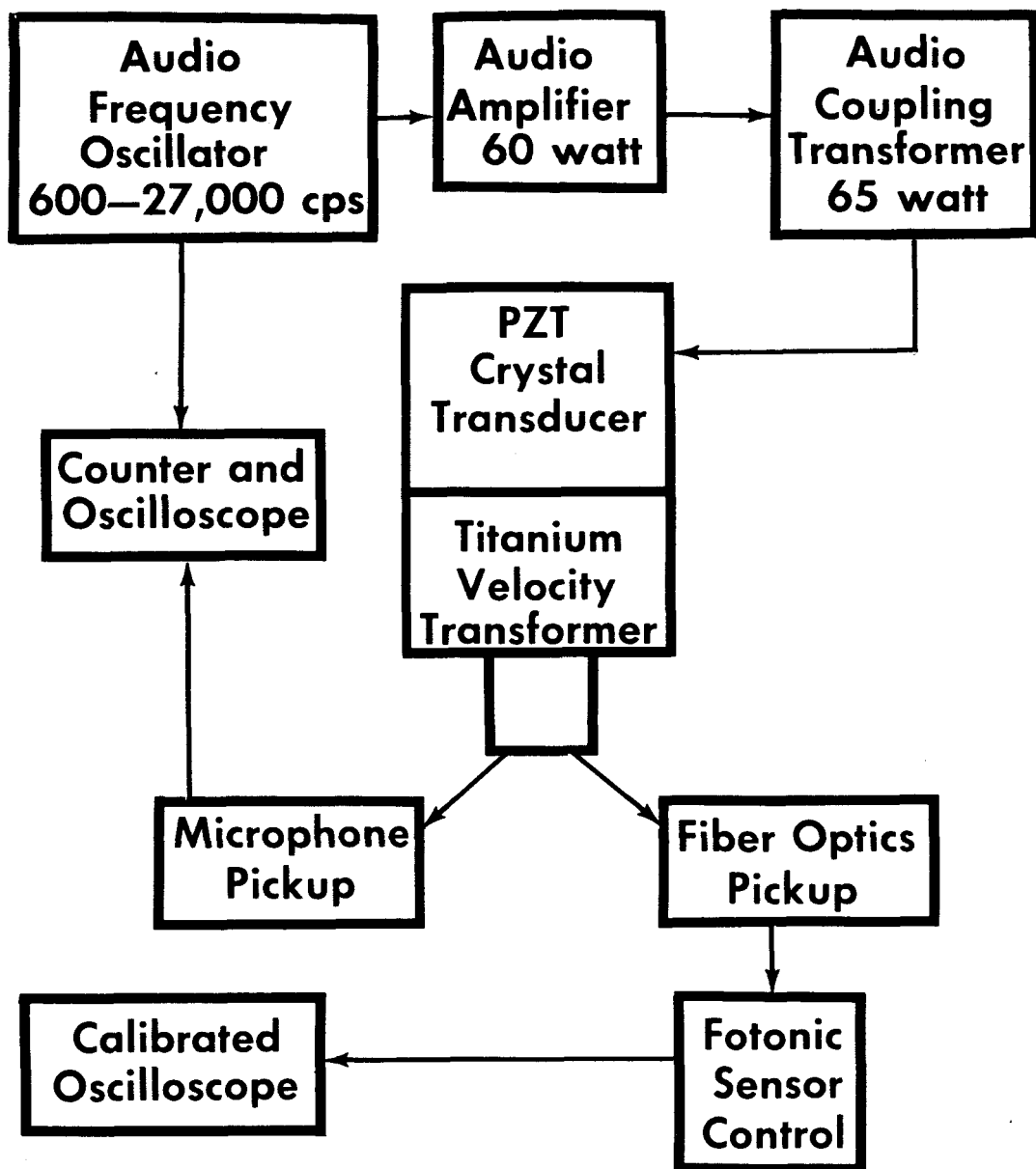
Bench Setup for Study of t he Vibration System

6370

FIG. 4

APPENDIX

CALCULATION OF THE RESONANT FREQUENCIES OF

CYLINDRICAL BARS AND VELOCITY TRANSFORMERS (STEPPED-HORNS)

The methods of calculating the resonant frequencies of simple resonant mechanical structures have been summarized by Neppiras[1]. For the extensional or longitudinal mode of vibration,

$$\ell_a = \frac{\pi}{\omega} \sqrt{\frac{E}{\rho}} \qquad\qquad \text{(a)}$$

where ℓ_a = the length of a cylindrical bar = $\frac{\lambda}{2}$,

λ = the wavelength of the resonant frequency in the material,

$\omega = 2\pi f$,

f = the resonant frequency,

E = the Young's modulus of the material, and

ρ = the density of the material.

Hence, the length of a vibrating cylindrical bar is determined by the material constants E and ρ and the desired resonant frequency f. In the case of a stepped-horn velocity transformer, Neppiras[2] gives the equation:

$$\ell_b = \frac{\pi}{2\omega} \sqrt{\frac{E}{\rho}} \qquad\qquad \text{(b)}$$

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[1]E. A. Neppiras; "Techniques and Equipment for Fatigue Testing at Very High Frequencies," ASTM Proc., Vol. 59 (1959), pp. 692.

[2]Ibid., p. 694.

where the ℓ_b in equation (b) is one-half ℓ_a in equation (a) and, therefore, equals $\lambda/4$. This "ℓ_b" is related to the geometry of the stepped-horn as shown below.
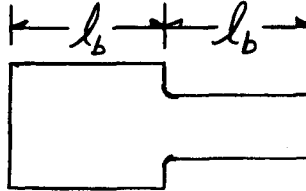
In the case of the titanium stepped horn, ℓ_b was calculated from equation (b) using the following data:

$$E = 11.6 \times 10^{11} \text{ dynes/cm}^{2(3)} = 16.8 \times 10^6 \text{ psi}$$

$$\rho = 4.50 \text{ gm/cm}^{3(4)}$$

$$f = 19.0 \times 10^3 \text{ cps} \quad \text{(assumed value)}$$

The length ℓ_b was 6.68 cm. (2.63 in.).

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[3] American Institute of Physics Handbook, McGraw-Hill, New York, 1957, Section 2, page 61.

[4] Ibid, Section 2, page 20.